

## **PERFORMANCE ANALYSIS OF SHELL AND TUBE HEAT EXCHANGERS USING AN EDUCATIONAL APPLICATION**

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### **Abstract**

The shell and tube heat exchangers are devices used very much in the industrial processes in many applications. These elements can have characteristics very special, in its use as well as in its design, therefore each system being a particular problem. The program developed uses simultaneously the LMTD and the effectiveness-NTU methods to such important heat transfer devices and may prove pedagogically useful to undergraduate mechanical engineering students attending heat transfer courses. The examples in this paper illustrate the power of the software developed in thermal sciences calculations with application to the heat-exchanger in configuration on line, staggered arrangement and of tube double. It is hoped that the software will bridge the gap between the

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teaching of engineering fundamentals and the existing industry practice of shell and tube heat exchanger design.

## 1. Introduction

The demand for educational software is growing exponentially with the surge of interest in educational reform, the internet, and distance learning. Educational applications must be very flexible because curricula and teaching styles vary tremendously among institutions, locations, and even among instructors at the same institution. In the classroom, educators and students have an investment in traditional monolithic applications that can bias them against adopting new technologies. A lot has been written about designing heat exchangers, and specifically, shell-and-tube heat exchangers. For example, the book by Kern [1] published in 1950 details basic design procedures for a variety of heat exchangers. Since the publication of that book, with the advent of computers, design procedures have become sophisticated even though the basic goals of design remain the same. This paper presents a didactical software developed for the design of shell and tube heat exchangers. Shell-and-tube heat exchangers (STHEs) are used extensively throughout the process industry and as such a basic understanding of their design, construction and performance is important to the practicing engineer. They are commonly used in process applications and in the air conditioning and refrigeration industry. They are also widely used as oil coolers, power condensers, preheaters and steam generators in both fossil fuel and nuclear-based energy production applications. They are also widely used. Although they are not specially compact, their robustness and shape make them well suited for high pressure operations.

The design of a heat exchanger involves initial conditions in which the following variables are known: (i) process-fluid rate of flow (ii) change in temperature of process fluid (iii) inlet temperature of utility fluid (for cooling or heating). With this information, the engineer must prepare a design for the optimum exchanger that will meet the required process conditions. Ordinarily, the following results must be determined: (i) heat-transfer area (ii) exit temperature and flow rate of utility fluid (iii) number, length, diameter, and arrangement of tubes (iv) tube-side and shell-side pressure drops and (v) efficiency.

To make the most of exchanger design software, one needs to understand shell-and-tube heat exchangers (STHEs) classification, exchanger components, tube layout, baffling, pressure drop, and mean temperature difference. However, a

previous good understanding of the underlying principles of exchanger design is needed to use the software effectively. The software called INTERCAM, was developed to be used in undergraduate classes of Mechanical and Chemical Engineering. The use of the software facilitates the development of individual or group projects, in which the situation can develop its creativity through the scanning of a real problem.

The software is suitable for teaching thermal design of shell and tube heat exchangers to senior undergraduate students in mechanical and chemical engineering and also to train new graduate engineers in thermal design. Students may be asked to use the software to undertake a few mini projects in shell and tube heat exchanger design. The software allows changing operation and design conditions such as temperature and flow inlet and outlet in shells and tubes, number and length of tubes, inside and outside diameter of tubes, number of passes, shell outside diameter, heat transfer coefficients in shells and tubes, heat transfer coefficient global, velocities of the flows and fouling factor. The three examples studied in this work illustrate how the software can be used in such a manner.

The present work shows the software structure developed for teaching of the shell and tubes heat exchangers. INTERCAM is very easy to use and user friendly. It is a software tool that is made to demonstrate the thermal design calculations of shell and tube heat exchangers. The program shows the static behavior of these devices under different design parameters and operations conditions. The software allows changing operation and design conditions such as temperature and flow inlet in shells and tubes, number and length of tubes, inside and outside fouling factor.

## **2. Development of the Software: INTERCAM**

The tutorial model has been developed to apply in the teaching of shell and tube heat-exchanger, programmed in C++. This software has main objective to solve problems to the heat-exchanger in configuration on line, staggered arrangement and of tube double. The software is divided in two sections. The first section is employed to introduce the data of the problem or charger previously a problem. The second section corresponds to the application to solve the problems.

The use of this software permits to the actors of the learning teaching process (students and teachers): to realize analysis and synthesis of the process that including in operations with heat-exchanger, to apply the concepts and theory knowledge of

various subjects in the practical of laboratory, to learn of manner autonomy and to incorporate new information and to decide the form more adequate use of it. The students can do practical in the laboratory with a real system and/or simulations in the software with data found in the literature. This methodology can be applied in second/third of a bachelor course. The software for education is a tool useful to get: class more active, problems with solutions different, programming of tasks of analysis and others formative actions. The use of this type of tools has been developed and proposed by various authors [2-6] developed a software structure for teaching and learning of the dynamics and control of shell and tubes heat exchangers. This software allows changing operation and design conditions such as temperature and flow inlet in shells and tubes, number and length of tubes, inside and outside diameter of tubes, number of passes, shell outside diameter and fouling factor. [7] introduced the effect of temperature on fouling into the design of heat exchangers, being quite complex because the final, design-fouling resistance depends on the history of operation of the exchanger. In the recent past, some experts studied on the design, performance analysis and simulation studies on heat exchangers [8-11].

It is possible to find two possibilities: development of educational software and use of commercial software. An overall skeleton of the menu, which is designed to be very user-friendly, is shown in Figure 1. There are three command boxes, namely Initial data, Configuration and Results. Initial data presents the values of the problem load. Configuration is possible to change the data, besides of design a new problem or to load other problem previously designed (Figure 2). Finally, the values calculated for all the parameters are shown in results. Figure 3 shows (see [12]) an example of a type of heat exchanger, a fixed-tube sheet heat exchanger. This device has straight tubes that are secured at both ends to tube sheets welded to the shell. The principal advantage of the fixed tube sheet construction is its low cost because of its simple construction. In fact, the fixed tube sheet is the least expensive construction type, as long as no expansion joint is required. Other advantages are that the tubes can be cleaned mechanically after removal of the channel cover or bonnet, and that leakage of the shell side fluid is minimized since there are no flanged joints. A disadvantage of this design is that since the bundle is fixed to the shell and cannot be removed, the outsides of the tubes cannot be cleaned mechanically.

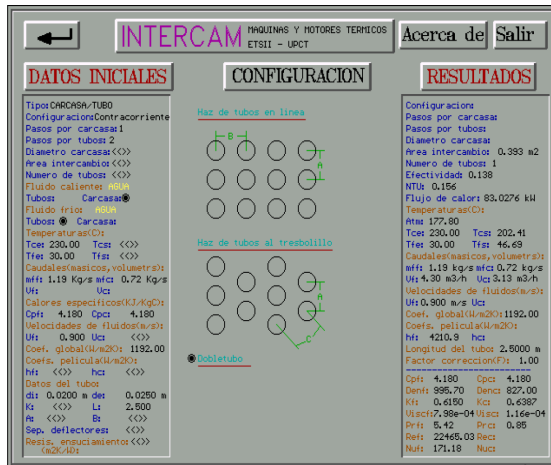


Figure 1. Main screen of the INTERCAM program.

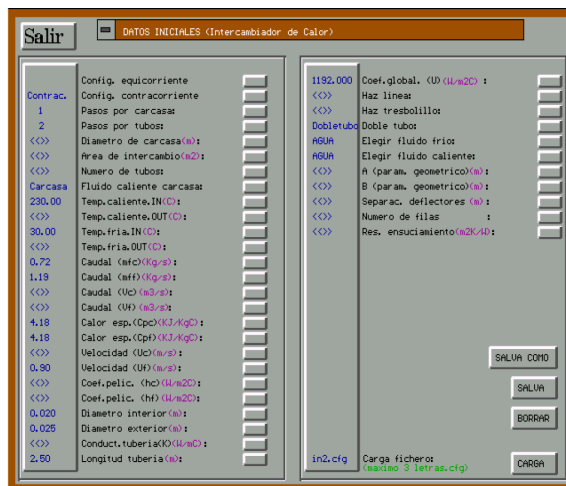


Figure 2. Screen of configuration of parameters.

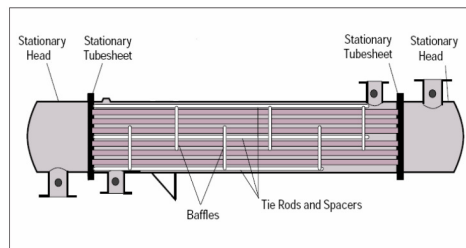


Figure 3. Example of shell-and-tube heat exchanger [8].

### 3. Educational Objectives

A heat exchanger is a device in which energy is transferred from one fluid to another across a solid surface. Exchanger analysis and design therefore involve both convection and conduction. Two important problems in heat exchanger analysis are (i) rating existing heat exchangers and (ii) sizing heat exchangers for a particular application. The shell and tube heat exchanger is equipment largely used at chemical industries. The developed software is intended to be used by undergraduate mechanical and chemical engineering students at the second year level. It can therefore be easily appreciated that the developed software has versatile teaching and educational uses.

The software can be a useful tool:

- (i) To introduce the student to the basic concepts of heat-exchanges.
- (ii) To enable the student to interactively create and utilize different types of dispositions of this type of heat-exchanger.
- (iii) To make the student appreciate the effects of various pertinent-actors such as global and convective heat transfer coefficients, efficiency, shell and tube passes, shell diameter, etc.
- (iv) To qualify the student to interpret the results obtained.
- (v) To enable the student to learn the fundamentals of this type of device.

### 4. Mathematical Model

The LMTD method can be readily used when the inlet and outlet temperatures of both the hot and cold fluids are known. When the outlet temperatures are not known, the LMTD can only be used in an iterative scheme. In this case the effectiveness-NTU method can be used to simplify the analysis. The basic heat exchanger equations applicable to shell and tube exchanger are shown here:

#### Total heat load transfer

$$\dot{q} = U \cdot A_o \cdot \Delta Tm_{cc} \cdot F, \quad (1)$$

where  $U$  is the global heat transfer referred to the area  $A_o$ , coefficient that depends that the heat transfer coefficients (inside and outside), geometry of the exchanger and thermal properties of both fluids.  $A$  is the heat transfer area,  $\Delta Tm_{cc}$  is the

logarithmic mean temperature difference for the pure countercurrent flow configuration and  $F$  is the configuration correction factor for multiple tube-side and/or shell-side passes.

$$U = \frac{1}{\frac{A_o}{A_i h_i} + \frac{A_o \ln(d_o/d_i)}{2\pi k L} + \frac{1}{h_o}}, \quad (2)$$

$$A_o = 2\pi r_o L, \quad (3)$$

$$\Delta T_{m_{cc}} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}}, \quad (4)$$

where  $T_{hi}$  and  $T_{ci}$  are the hot and cold inlet temperatures, respectively,  $T_{ho}$  and  $T_{co}$  are the corresponding outlet temperatures.

The value of  $F$  depends upon the exact arrangement of the streams within the exchangers, the number of exchangers in series, and two dimensionless parameters defined in terms of the terminal temperatures of the two streams:

$$R = \frac{(T_{hi} - T_{ho})}{(T_{co} - T_{ci})}, \quad (5)$$

$$P = \frac{(T_{co} - T_{ci})}{(T_{ho} - T_{ci})}. \quad (6)$$

The mathematical relationships between  $R$ ,  $P$  and  $F$  can be found in numerous books [1, 13], we have considered in this software the configurations more used, 1-2 and 2-4. Values of  $F$  below 0.75 at the lowest should not be used, because it means that substantial additional area must be supplied in the heat exchanger to overcome the inefficient thermal profile.  $F$  is unity for the cases of purely parallel and countercurrent flow. The Arithmetic Mean Temperature Difference is the following in this case.

$$\Delta T_{m_p} = \frac{(T_{hi} - T_{ci}) - (T_{ho} - T_{co})}{\ln \frac{(T_{hi} - T_{ci})}{(T_{ho} - T_{co})}}. \quad (7)$$

The expression of  $\Delta T_m$  for the case purely concurrent (parallel) is rarely used. The LMTD method and NTU method are used in the software developed. The NTU method is an alternative to calculate the total heat transfer rate. This method brings

out the individual effects of not only the total thermal conductance  $UA$  but also that the capacity rates  $C_h$  and  $C_c$ , so appears a new dimensionless number, denominated number of heat transfer units:

$$NTU = UA / C_{\min}, \quad (8)$$

where  $C_{\min}$  is the smaller of the two capacity rates.

Other dimensionless number is the heat exchanger effectiveness  $\varepsilon$ , which is defined as the ratio between the actual heat transfer rate  $\dot{q}$  and the maximum heat transfer that could take place in the heat exchanger ( $\dot{q}_{\max} = C_{\min}\Delta T_{\max}$ ).

$$\varepsilon = \dot{q} / \dot{q}_{\max}. \quad (9)$$

### Heat-transfer coefficient

The tube side heat-transfer coefficient is a function of the physical properties (viscosity, thermal conductivity and specific heat), the tube diameter and very importantly, mass velocity. It is necessarily calculated the Reynolds number and the Prandtl number. The variation in liquid viscosity is quite considerable; so, this physical property has the most dramatic effect on heat-transfer coefficient. The fundamental equation for turbulent heat-transfer inside tubes is:

$$Nu = 0.027Re^{0.8}Pr^{0.33}. \quad (10)$$

The models derived above are simulated using the software developed in this work and called INTERCAM. To built this program, it was necessary to use the programming Language C++.

## 4. Results and Discussions

Usually, the flow rates and the physical properties of the two streams involved are specified, and the temperatures at which the fluids are available are known. If the outgoing temperature of one of the streams is not specified, usually a constraint (e.g., the temperature of the cooling water cannot exceed 99°C) is given. Then, by an energy balance, the outgoing temperature of the second stream can be calculated along with the heat duty. The heat duty  $\dot{q}$  is usually fixed by the required service. The tube diameter, tube length, shell types, etc. are all standardized and are available only in certain sizes and geometry. The selected heat exchanger has to meet or



exceed this requirement.

### Problem I

Configuration: 1-2 (one tube pass and two shell passes) and 2-4 (two tube passes and four shell passes), double tube.

Hot fluid (shell side): water /air NH<sub>3</sub> /oil, 1.19 kg/s,  $T_{hi} = 230^{\circ}\text{C}$ .

Cold fluid (tube side): water, 0.72 kg/s,  $T_{ci} = 30^{\circ}\text{C}$ .

Global heat transfer coefficient:  $U = 1192 \text{ W/m}^2\text{K}$ .

Geometric parameters:  $d_i = 40 \text{ mm}$ .

Length of the tube: 2.5 m.

**Table 1.** Problem I. Configuration 1-2, double tube for various hot fluids

Hot fluid	$\dot{q}$ (kW)	$c_{e,h}$ (kJ/ KgK)	$NTU$	$\epsilon$	$\Delta Tm$
Water	124.12	4.180	0.249	0.206	166.77
Air	89.96	1.031	1.009	0.6065	123.25
NH <sub>3</sub>	114.98	2.467	0.422	0.324	155.15
Oil	125.64	4.689	0.222	0.186	168.70

The effect of different input variables on output variable are discussed in detail in the following cases.

Problem I is studied in the Tables 1-3. Tables 1 and 2 show the effect of the change of the hot fluid (water, air, NH<sub>3</sub> and oil) for the configurations 1-2 and 2-4, respectively. When the hot fluid chosen is the air, the biggest values of NTU and  $\epsilon$  are obtained, however  $\dot{q}$  decreases. We can see that the best configuration is the 2-4, because for this configuration are reached the performance maximum.

**Table 2.** Problem I. Configuration 2-4, double tube. Effects of the hot fluid

Hot fluid	$\dot{q}$ (kW)	$c_{e,h}$ (kJ/ KgK)	$NTU$	$\epsilon$	$\Delta Tm$
Water	326.29	4.180	0.498	0.542	111.63
Air	142.65	1.031	2.018	0.961	52.98
NH <sub>3</sub>	262.86	2.467	0.483	0.843	91.48
Oil	337.81	4.689	0.444	0.500	115.27

Table 3 shows the effects of inlet hot fluid temperature. It can be seen that an increase in the  $T_{hi}$  leads to a decrease in heat exchanger effectiveness and NTU. Besides, the actual heat transfer rate increases as the inside hot fluid temperature increases.

**Table 3.** Problem I. Configuration 1-2, double tube. Effects of inlet hot fluid temperature

$T_{hi}$ (°C)	$\epsilon$	$NTU$	$\dot{q}$ (kW)	$\Delta T_m$
220	0.607	1.011	85.39	117.01
200	0.608	1.015	76.29	104.54
180	0.609	1.018	67.23	92.14
160	0.610	1.021	58.20	79.76

### Problem II

Configuration: in-line tube banks (Figure 4a).

Hot fluid (shell side): water, 3.55 kg/s,  $T_{hi} = 120^\circ\text{C}$ ,  $u_h = 0.5$  m/s.

Cold fluid (tube side): oil, 5 kg/s,  $u_c = 0.4$  m/s,  $T_{ci} = 30^\circ\text{C}$ ,  $T_{co} = 85^\circ\text{C}$ .

Global heat transfer coefficient:  $U = 1550$  W/m<sup>2</sup>K.

Geometric parameters:  $d_i = 20$  mm,  $d_o = 25$  mm.

With this parameters are obtained the following constant values for all the cases studied:  $N = 46$ ,  $\epsilon = 0.611$ ,  $NTU = 1.630$ ,  $c_{pf} = 2$  kJ/kgK,  $c_{pc} = 4.18$  kJ/kgK,  $\dot{q} = 550$  kW. Problem II is studied in the Tables 4 and 5. In Table 4 is used the 1-2 configuration, while in Table 5 is used the 2-4 configuration. In this case the effect of the global heat transfer coefficient is analyzed. An increase of this parameter leads to a decrease in the heat transfer and of the length of the tubes, which was expected. However,  $\epsilon$  and NTU does not change. As in the previous case, the best configuration is the 2-4, because for this configuration are obtained less values of the heat transfer area and of the length of the tubes with the same performance.

**Table 4.** Problem II. Configuration 1-2, in-line tube banks. Effects of the global heat transfer coefficient

$U$ (W/m <sup>2</sup> K)	$A$ (m <sup>2</sup> )	$L$ (m)	$\epsilon$	$NTU$
1250	13.039	1.8045	0.611	1.630
1350	12.073	1.6708	0.611	1.630
1450	11.240	1.5556	0.611	1.630
1550	10.515	1.4552	0.611	1.630

**Table 5.** Problem II. Configuration 2-4, in-line tube banks. Effects of the global heat transfer coefficient

$U$ (W/m <sup>2</sup> K)	$A$ (m <sup>2</sup> )	$L$ (m)	$\epsilon$	$NTU$
1250	10.651	0.7370	0.767	1.331
1350	9.862	0.6824	0.767	1.331
1450	9.182	0.6354	0.767	1.331
1550	8.589	0.5944	0.767	1.331

**Problem III**

Configurations: 1-2, in-line and staggered tube banks (Figure 4).

Hot fluid (shell side): water, 3.3 kg/s,  $T_{hi} = 100^\circ\text{C}$ ,  $T_{ho} = 70^\circ\text{C}$ .

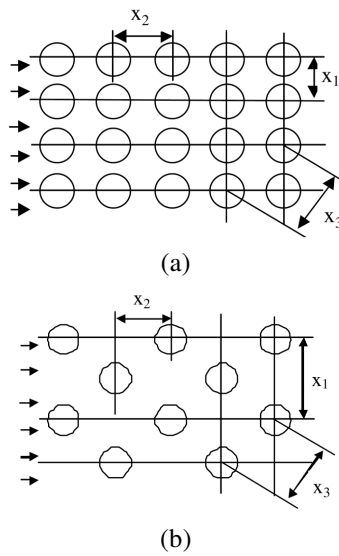
Cold fluid (tube side): water, 5.1 kg/s, 0.3 m/s,  $T_{ci} = 35^\circ\text{C}$ .

Thermal conductivity of the tube:  $k = 100 \text{ W/mK}$ .

Geometric parameters:  $d_i = 20 \text{ mm}$ ,  $d_o = 25 \text{ mm}$ .

Baffle spacing:  $S_b = 0.2 \text{ m}$ .

$X_1 = 0.075 \text{ m}$ ,  $X_2 = 0.05 \text{ m}$ .



**Figure 4.** Configurations: (a) Configuration in-line tube banks, (b) Configuration staggered tube banks.

**Table 6.** Problem III. Configuration in-line tube banks. Effects of the distance between deflectors

$S_b$ (m)	$h_c$ ( W/mK )	$L$ (m)	$A$ (m <sup>2</sup> )	$\Phi_s$ (m)	$U$ (W/m <sup>2</sup> K)
0.10	1886.9	1.319	11.396	0.3725	968.4
0.15	1461.5	1.516	13.098	0.4169	842.5
0.20	1219.3	1.689	14.598	0.4570	755.9
0.25	1059.4	1.848	15.964	0.4938	691.2
0.30	949.4	1.994	17.232	0.5281	640.4

The effect of the baffle spacing is analyzed in Table 6. It can be appreciated that when  $S_b$  increases,  $A$ ,  $L$  and the shell diameter ( $\Phi_s$ ) increases, while  $h_c$  (heat transfer coefficient for cold fluid or tube side) and  $U$  (global heat transfer coefficient) decreases. In Table 7 is studied the effect of the number and distribution (column  $\times$  row) of the tubes. It can be seen that an increase of  $N$  deals a decrease of the velocity, of the length tube and of the global heat transfer coefficient. The shell diameter ( $\Phi_s$ ) increases when  $N$  increases, however,  $\Phi_s$  is 0.4773 and 0.4718 for  $N = 60$  (12 columns  $\times$  5 rows) and  $N = 66$  (11 columns  $\times$  6 rows), respectively.

**Table 7.** Problem III. Configuration staggered tube banks. Effects of the number and distribution of the tubes

$N$	$u_f$ ( m/s )	$L$ (m)	$A$ (m <sup>2</sup> )	$\Phi_s$ (m)	$U$ (W/m <sup>2</sup> K)
50(10 $\times$ 5)	0.328	1.6337	12.831	0.4357	860.0
60(10 $\times$ 6)	0.273	1.4475	13.643	0.4498	808.9
60(12 $\times$ 5)	0.273	1.5415	14.529	0.4773	759.6
66(11 $\times$ 6)	0.248	1.4051	14.568	0.4718	757.5

Tables 8 and 9 show the effect of  $X_1$  (vertical distance between tubes) for the configurations in-line and staggered tube banks, respectively. It can be seen that an increase of  $X_1$  leads to a decrease in global heat transfer coefficient ( $U$ ) and heat transfer coefficient  $h_c$ . Besides,  $L$  and  $A$  increase when  $X_1$  increases. We can see that for the staggered tube banks configuration are obtained bigger values of  $U$  and  $h_c$  than the in-line tube banks configuration. Besides, the values of  $L$  and  $A$  are smaller, therefore the staggered tube banks configuration is the most adequate.

**Table 8.** Problem III. Configuration in-line tube banks. Effects of the vertical distance between tubes

$X_1$ (m)	$h_c$ (W/mK)	$L$ (m)	$A$ (m <sup>2</sup> )	$\Phi_s$ (m)	$U$ (W/m <sup>2</sup> K)
0.070	1302.9	1.6224	14.017	0.4414	787.3
0.075	1219.3	1.6897	14.598	0.4720	755.9
0.080	1148.2	1.7546	15.158	0.4570	728.0

**Table 9.** Problem III. Configuration staggered tube banks. Effects of the vertical distance between tubes

$X_1$ (m)	$h_c$ (W/mK)	$L$ (m)	$A$ (m <sup>2</sup> )	$\Phi_s$ (m)	$U$ (W/m <sup>2</sup> K)
0.070	1428.7	1.5362	13.271	0.4414	831.5
0.075	1359.8	1.5815	13.663	0.4720	807.7
0.080	1300.9	1.6240	15.158	0.4570	786.5

**Table 10.** Problem III. Configuration in-line tube banks. Effects of the inlet cold fluid temperature

$T_{ci}$ (°C)	$F$	$L$ (m)	$NTU$	$\epsilon$	$U$ (W/m <sup>2</sup> K)
25	0.96	1.3618	0.624	0.400	731.2
30	0.95	1.5029	0.700	0.429	744.0
35	0.94	1.6897	0.800	0.462	755.9
40	0.91	1.9497	0.910	0.500	766.8
45	0.88	2.3366	1.137	0.545	777.0
50	0.81	2.9461	1.477	0.600	786.4

Finally, Table 10 shows the effect of  $T_{ci}$  (inlet cold fluid temperature) in the configurations in-line. It can be observed that an increase in  $T_{ci}$  leads to an increase in the heat transfer efficiency ( $\epsilon$ ), global heat transfer coefficient ( $U$ ), transmission unities number ( $NTU$ ) and the length tube ( $L$ ), while the LMTD correction factor ( $F$ ) decreases when  $T_{ci}$  increases.

### Conclusion

This software has been developed for the thermal design of shell and tube heat exchangers based on the LMTD and efficiency methods ( $NTU$ ). The software is highly user-friendly and has many capabilities to create new problems in the field of the mechanical engineering in the design of shell-tube heat-exchanger. It is hoped that the software will bridge the gap between the teaching of fundamentals and industry practice of this type of device. The software incorporate various design

options for the heat exchangers including the variations in the tube diameter, tube pitch, shell diameter, number of tube passes, baffle spacing, baffle cut, etc. Three configurations can be used, on line, staggered arrangement and of tube double. The software was tested using some thermal problems and found to provide results that are very accurate. Trial tests show that INTERCAM solves the problems directly and effectively using an ordinary PC.

### Nomenclature

$A$	[m <sup>2</sup> ] Heat-transfer area
$c_e$	[kJ/kgK] Specific heat
$d_i$	[m] Tube inside diameter
$d_o$	[m] Tube outside diameter
$F$	[-] LMTD correction factor
$h$	[W/m <sup>2</sup> K] Heat-transfer coefficient
$k$	[W/mK] Thermal conductivity
$L$	[m] Length tube
$\dot{q}$	[KW] Heat transfer rate
$\dot{m}$	[kg/s] Flow rate
$N$	[-] Number of tubes
$NTU$	[-] Transmission unities number
$Nu$	[-] Nusselt number
$Pr$	[-] Prandtl number
$P$	[-] Dimensionless parameter, LMTD
$R$	[-] Dimensionless parameter, LMTD
$R_f$	[m <sup>2</sup> K/kJ] Fouling resistance

- $Re$  [-] Reynolds number  
 $T$  [°C] Temperature  
 $u$  [m/s] Velocity  
 $U$  [W/m<sup>2</sup>K] Global heat transfer coefficient  
 $S_b$  [m] Baffle spacing

### Special characters

- $\varepsilon$  [-] Efficiency  
 $\mu$  [kg/m s] Viscosity  
 $\Phi_s$  [m] Shell diameter

### Subscripts

- $c$  Cold  
 $h$  Hot  
 $i$  Inlet  
 $o$  Outlet  
 $s$  Shell  
 $t$  Tube

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